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Compression device for gaseous media

The present invention relates to a device for compressing gaseous media. The invention is described with regard to a compressor, in particular for an air-conditioning system of a motor vehicle; however, it is pointed out that the device may also be used for other devices for compressing gaseous media.

Such compressors are known from the prior art for air-conditioning equipment as a basic component of the same. Likewise, it is known from the prior art that the compressors or refrigeration compressors constitute a large loss source for the air-conditioning system inasmuch as they lead to a significant, additional energy consumption and thus fuel consumption.

The causes of these energetic losses are different irreversibilities which on the one hand increase the compressor driving power and on the other hand increase the thermal output which is to be dissipated to the environment. Depending on the refrigerant used, the losses in the compressor split up in different ways into various loss processes. The most important loss processes include the stroke-independent friction power, the stroke-dependent friction power, the power loss due to internal leakage, the throttling losses at the suction valve and the throttling losses at the pressure valve. Opposed effects on the individual loss processes can be produced by design changes to the construction of the compressor.

For example, measures for improving the inner tightness between a volumetric displacement means, in particular a piston, and the wall assigned to it, in particular a cylinder wall, are at the same time reflected in an increase in the friction power, which again nullifies some of the improvements.

Intensive investigations have been able to show that in particular throttling losses at the suction valve have an adverse effect, in particular at high suction volumetric flows delivered, that is to say at a high speed and at a high delivery efficiency of the compressor, and when using a refrigerant having a relatively low volumetric refrigerating capacity, such as R134a for example.

The object of the present invention therefore consists in improving the overall efficiency of a compressor, in particular at high volumetric flows, by the pressure loss at the suction valve being reduced by design measures. This is achieved according to the invention by a device as claimed in claim 1. Advantageous developments and embodiments are the subject matter of the subclaims.

The device according to the invention for compressing gaseous media has at least one compression space into which the gaseous medium can enter and from which the gaseous medium can discharge. Furthermore, at least one first valve means having at least one first opening and at least one first covering means essentially covering the first opening at least intermittently is provided, the first valve means allowing the gaseous medium to enter the compression space and essentially preventing a

discharge of the gaseous medium from the compression space.

In addition, a second valve means having at least one second opening and at least one second covering means essentially covering the second opening at least intermittently is provided, the second valve means allowing a discharge of the gaseous medium from the compression space and essentially preventing the gaseous medium from entering the compression space.

According to the invention, the free cross section of one valve means considerably exceeds the free cross section of the other valve means.

The expression "the free cross section" or "the narrowest free cross section" refers to the surface or the lateral area, peripherally defining the opening, of the geometrical space or volume whose height is defined by the distance of the covering means (with open valve) from the opening and whose periphery is defined by the periphery of the opening cross section of the valve. In this case, the distance between the covering means and the opening need not necessarily be constant.

The free cross section of the first valve means considerably exceeds the free cross section of the second valve means. This means that the free cross section of the valve means which enables the gaseous medium to be drawn into the compression space considerably exceeds the free cross section of the valve means which enables the gaseous medium to be discharged from the compression

space. The corresponding cross sections at the suction valve are thus designed to be larger than the cross sections at the pressure valve of the compression device.

The expression "covering means" in this case refers to a means which essentially completely covers, at least intermittently, the opening assigned to it and therefore acts in a sealing manner for the opening in this state.

In a further preferred embodiment, the free cross section of the one valve means exceeds the free cross section of the other valve means at least by a factor of 2; this means that the free cross section of the suction valve exceeds the free cross section of the pressure valve by at least a factor of 2. The free cross section of the one valve means preferably exceeds the free cross section of the other valve means at least by a factor of 2.5, preferably at least by a factor of 3, and in particular preferably at least by a factor of 4.

In a further preferred embodiment, the device has a piston means arranged so as to be movable relative to the compression space, one respective valve closing and one being opened at least intermittently as a function of the direction of movement of the piston.

In a further preferred embodiment, at least one covering means is designed as a reed. Both covering means are preferably designed as reeds. The latter, relative to the compression space, depending on the direction of movement of the piston means, are either at a distance from the openings assigned to them or bear essentially against

said openings, so that in this way the passage of gas in one direction through the opening is prevented and is essentially permitted in the other direction.

In a further preferred embodiment, at least one valve means is arranged in a valve plate, and preferably both valve means are arranged in said valve plate. The valve plate forms the closure of the compression space. Thus the above-designated distance between the opening and the covering means also refers to the distance between the valve plate or that surface of the valve plate which faces the covering means, on the one hand, and the covering means, on the other hand.

In a further preferred embodiment, the first opening of the first valve means, i.e. the suction valve means, is designed to be noncircular.

As explained above, the free cross section of the valve means results from the periphery of the valve opening cross section and the distance between the opening and the covering means. In this embodiment, therefore, by selecting a noncircular cross section, the periphery of the opening is increased at the same cross-sectional area or lateral dimension and thus with the same space requirement. It is known that a circle has the smallest ratio between circle periphery and circle area compared with other two-dimensional geometrical figures. The modification of the opening cross section from the circular profile therefore produces an increase in the ratio of periphery and area of the opening. In other words, a ratio between circle periphery and circle area

is selected which is greater than 2/r, where r is the radius of the circle opening.

The advantage of this procedure is that the periphery of the opening can be increased without at the same time the cross section or the area of the opening increasing to a comparable degree, as a result of which the areas available only to a limited extent on the valve plate can be taken into account.

In a further preferred embodiment, the first valve means has a plurality of first openings. In this way, too, the periphery of the valve opening cross section in relation to its area can be increased.

In a further preferred embodiment, the periphery of the at least one first opening of the first valve means is greater than, preferably markedly greater than, the periphery of the at least one second opening of the second valve means. This means that the suction valve has a larger, preferably considerably larger, opening periphery than the pressure valve. In this way, as mentioned at the beginning, the loss of the device for compressing gas can be considerably reduced, even if an increase in slight losses at the pressure valve are tolerated in the process.

In a further preferred embodiment, the at least one opening of the first valve means, compared with an imaginary circular opening which has the same cross section as the at least one opening, has a periphery which exceeds the periphery of said imaginary circular

opening by at least 10%, preferably by at least 20% and in particular preferably by at least 50%. This means that the actual opening is compared with an imaginary circular opening, the imaginary opening having the same cross-sectional area as the actual opening, and the actual opening on the other hand having a larger periphery than the imaginary opening. As explained above, this can be achieved, for example, by deviations, preferably significant deviations, from the circular shape.

The specified increases in the periphery/area ratio with regard to the circular opening by 10%, 20% or 50%, which are also considered to be significant increases, produce a considerable reduction in the power loss at the suction valve.

In a further preferred embodiment, at least one covering means, preferably the covering means of the first valve means, has at least one aperture. The expression "aperture" in this case refers to an interruption in the covering means. Here, the aperture may have any desired geometrical shapes, for example circular, elliptical, polygonal and/or similar cross sections. In this case, the apertures may be arranged in the regions where a long gas path of narrow cross section would be produced if these apertures were not present.

In a further preferred embodiment, at least one covering means, preferably the first covering means of the first valve means, has at least one projection. In contrast to the abovementioned aperture, a recess refers to a formation which projects from the remaining area, whereas

the aperture is surrounded essentially over the full periphery by the area of the covering means.

In a further preferred embodiment, at least one covering means is fastened to the valve plate; both covering means are preferably fastened to the valve plate.

In a further preferred embodiment, the configuration of at least one covering means is adapted to the configuration of the opening assigned to this covering means. The expression "opening assigned to this covering means" refers to that opening which the relevant covering means is intended to cover. The peripheral margins of at least one covering means preferably project beyond the peripheral margins of the associated opening by between 0.5 mm and 5 mm, preferably by 1 mm to 3 mm. This means that, if, for example, the opening were to be of circular design with a radius of 20 mm, the covering means assigned to it would be arranged concentrically relative to the opening with a radius of between 20.5 mm and 25 mm, preferably between 21 mm and 23 mm. In this case, the covering means can project by an essentially constant amount along the entire periphery beyond the opening assigned to it; however, the amount of overlapping may also vary, so that the covering means projects beyond the opening to a different extent at different regions.

In a further preferred embodiment, at least one opening has a groove encircling the opening cross section. The abovementioned slight overlap has the advantage that the damping when the covering means lifts and comes into contact due to the breakaway or the displacement of the

gas cushion and/or refrigerating-oil cushion in the narrowest gap is minimized. In order to minimize this effect, a groove encircling the opening cross section or specific roughening of the valve plate may be additionally provided.

In a further preferred embodiment, the valve plate, preferably on the side facing the covering means, has at least one surface section having a coating which is deformable at least in sections.

In a further preferred embodiment, at least one covering means has at least one surface section, preferably on the side facing the opening, having a coating which is deformable at least in sections. In this case, in an especially preferred embodiment, the coating has at least one material which contains Teflon (PTFE).

The reason for this embodiment is that, due to the significant increase in the periphery of the suction valve opening and of the valve plate, the sealing area between the valve plate, on the one hand, and the covering means, on the other hand, is also increased and thus additional leakage cross sections may arise. Due to the elastically and/or plastically deformable coating on the valve plate and/or the covering means, these leakages can at least be reduced. As stated, temperature-resistant polymers, such as Teflon (PTFE), are suitable for the coating, but metallically soft claddings are also able to compensate for microroughness by plastic adaptation of the sealing members. In the latter case, however, it is necessary to fix the covering means or reeds in position

with respect to the cylinder base, which, however, does not pose a technical problem.

In a further preferred embodiment, in the open state of the valve, at least a section of at least one covering means is at a distance from its assigned opening which is greater than 0.5 mm, preferably greater than 1.0 mm, and in particular greater than 1.5 mm. This preferably involves the covering means of the first valve means.

As explained above, the narrowest free cross section of the suction valve is to be increased, this cross section resulting from the product of the periphery of the valve opening and the distance of the covering means from the opening or the valve plate. Instead of the periphery of the opening being increased, the distance can therefore also be increased. However, an increase in this distance also leads to the opening and closing times of the valve being increased and as a result additional internal leakages may occur due to the valve closing too late. However, it is possible to improve an increase of the maximum permitted stroke of the covering means relative to the valve plate in connection with an adaptation of the spring rigidity and/or prestress of the covering means.

The invention also relates to the use of the device according to the invention in an air-conditioning system, in particular for a motor vehicle. However, it should be made clear that such devices for compressing gas may also be used in other refrigerating machines, such as domestic refrigerators for example.

Further advantages and embodiments of the present invention follow from the attached drawings, in which:

fig. 1 shows a plan view of a valve plate of a device for compressing a gaseous medium according to the prior art;

fig. 2 shows a schematic section through the valve plate of the device from fig. 1;

fig. 3 shows a diagram for illustrating the compressor power losses in a device according to the prior art;

fig. 4 shows a plan view of a first embodiment according to the invention of a valve plate of a device for compressing a gaseous medium;

fig. 5 shows a plan view of the device according to the invention in a further embodiment;

fig. 6 shows a plan view of a device according to the invention in a further embodiment;

fig. 7 shows a plan view of a device according to the invention in a further embodiment;

fig. 8 shows an illustration of the power losses for the device according to the invention;

fig. 9 shows an illustration of the efficiency for a

device according to the prior art; and

fig. 10 shows an illustration of the efficiency for a device according to the invention for compressing a gaseous medium.

Fig. 1 shows a plan view of the piston-side surface of a valve plate 2 of a compression device according to the prior art. Provided in this valve plate is a pressure valve opening 4 which is provided with a covering means (not shown). Also shown is a covering means 7, a second (concealed) valve opening 13, which is part of the suction valve.

In the compression device according to the invention, both the opening 4 of the pressure valve and the opening 13 of the suction valve have peripheries of similar size.

Fig. 2 shows a schematic detail of a compression device according to the prior art. In this case, the designation 7 designates the covering means of the suction valve, which is open in this state, the bottom end 7a of the suction valve reed coming to lie at the left-hand stop of the notch 3 and thus being prevented from moving further away from the opening 13. The top end 7b of the covering means of the suction valve is fastened between the valve plate 2 and a cylinder wall 18. The designation 13 relates to the opening of the suction valve, this opening being essentially covered by the covering means 7 in the closed state. The designation 4 identifies the opening of the pressure valve, this opening likewise being essentially covered by the covering means 8 in the closed

state shown here. The covering means 8 is fastened with the bottom end between the valve plate 2 and a separating web 14. This separating web 14 serves to seal off the suction space 12 from the pressure space 11 in an essentially gas-tight and/or liquid-tight manner. The compression space 10 or its end region is closed off by the cylinder wall 18 and the valve plate 2. A piston means (not shown) moves inside the compression space 10, either the suction valve or the pressure valve being closed, depending on the direction of movement of the piston device. The designations 16a and 16b designate the annular grooves around the respective valve openings 13 and 4. These annular grooves serve to minimize the time delays when the respective valve covering means lift or come into contact due to the breakaway or the displacement of the gas cushion and/or refrigerating-oil cushion.

An illustration of the compressor power losses for a compression device according to the prior art is shown in fig. 3. This is based on defined pressures and on the original suction valve at a high pressure ratio. The total power losses in relation to the isentropic power, that is to say the power at constant entropy, is shown in the y axis. Different respective compressor speeds in the unit rev/min is shown in the x axis.

In this case, the loss proportions relate to the isentropic work of compression, that is to say the loss-free work of compression. The four diagrams show the respective power losses at different delivery efficiencies, the designation A identifying a delivery

efficiency of 0.8, the designation B identifying a delivery efficiency of 0.6, the designation C identifying a delivery efficiency of 0.4 and the designation D identifying a delivery efficiency of 0.2.

Here, the delivery efficiency is defined as the product of the control efficiency and the volumetric efficiency. In this case, the control efficiency $\lambda_{control}$ is defined as follows as the ratio of the current geometric swept volume of the compressor controllable in power output and the maximum geometric swept volume:

$$\lambda_{control} = V_{geo} / V_{geo-max}$$

The volumetric efficiency is defined in conventional manner as the ratio of the actually delivered volumetric flow relative to the volumetric flow theoretically delivered at the current swept volume, according to the following equation:

$$\lambda_{volumetric} = G_R / (\rho_{suction} \cdot V_{geo} \cdot r_c)$$

For the delivery efficiency, that is to say the product of the volumetric efficiency and the control efficiency, the follow ratio is thus obtained:

$$\lambda_{delivery} = G_R / (\rho_{suction} \cdot V_{geo-max} \cdot r_c)$$

The quantitative determination of the individual loss contributions to the total loss is complicated and requires extensive measuring at the compressor, for

example indication of the compression behavior by means of a high definition measuring technique.

In this case, compressor optimization which takes place by "trial and error" is known from the prior art, only the effect on the efficiency, which is easy to determine, and the volumetric efficiency actually being determined. The way in which the individual loss contributions are quantitatively split up under the various operating conditions is not determined according to the prior art. Such a resolution leads to an optimization potential for the compressors which can be improved even further.

Thus, as mentioned at the beginning, design changes to the compressor configuration possibly lead to opposed effects on the individual loss processes, in which case the magnitude of these effects may provide important hints for further optimization steps.

However, the individual loss proportions can be quantified by means of a computer analysis of measured data, as a result of which conclusions can be drawn with regard to the dominating loss mechanisms under various basic or operating conditions. However, this optimization process is not the subject matter of the present invention and is consequently not described in more detail. On the contrary, the result of such a loss analysis is explained with reference to an actual example and the conclusions are drawn which can result in an increase in the overall efficiency in a R1349 compressor for a motor-vehicle air-conditioning system.

In this case, the designation 31a in fig. 3 relates to the relative stroke-independent friction power, the designation 31b relates to the relative stroke-dependent friction power, the designation 31c relates to the relative leakage loss, the designation 31d relates to the relative pressure valve loss and the designation 31e relates to the relative suction valve loss. It can be seen that the relative stroke-independent friction power and the relative stroke-dependent friction power are essentially independent of the respective compressor speed. The relative leakage loss 31c and the relative pressure valve loss 31d change as a function of the compressor speed. In particular at high delivery efficiencies, as shown in figs A and B, it can be seen that the relative suction valve loss 31e greatly increases as a function of the compressor speed toward high compressor speeds and, in particular at high delivery efficiencies and high compressor speeds, the relative suction valve loss 31e dominates the overall power loss.

For this reason, the total power loss can be considerably reduced, in particular at high compressor speeds, by a reduction in the relative suction valve loss. In actual operating states of an air-conditioning compressor, the suction power through the valve gap of the suction valve reaches values of over 1000 W.

By a reduction in these losses at the suction valve, the total loss can thus be reduced even if the pressure valve loss, which has a less pronounced effect on the total loss compared with the suction valve loss, is increased

by the same measure.

A device according to the invention for compressing gaseous media is shown in a first embodiment in fig. 4. This device has one pressure valve opening 4 and two suction valve openings 13a and 13b. The result of this is that the periphery of the suction valve openings far exceeds the periphery of the pressure valve opening 4, i.e. it is essentially twice the size in this exemplary embodiment. The designation 7 identifies the covering means of the suction valve, this covering means completely covering the valve openings 13a and 13b in the closed state.

As mentioned at the beginning, the largest proportion of the pressure loss is produced at the narrowest cross section of the respective valve. In the conventional design of the compressor valves, this is generally the lateral area of the column-like structure (cf. fig. 2), the height of which is defined by the distance of the covering means from the valve plate and the periphery of which is defined by the periphery of the valve opening cross section of the valve plate. This means that the narrowest free cross section of the suction valve in relation to the narrowest cross section of the pressure valve is defined by the respective defined lateral areas in the product with the distances of the covering means from the valve plate.

A further embodiment of a device according to the invention is shown in fig. 5. In this embodiment, the considerably larger periphery of the suction valve

relative to the pressure valve is achieved by this valve opening having a substantially larger circular cross section. It is to be noted in this case, however, that a sufficiently wide web (not shown) remains between the respective openings for the pressure valve and the suction valve, this web permitting separation between the pressure space and the suction space on that side of the valve plate which is opposite the cylinder, in which or from which the gas to be compressed flows (cf. fig. 2).

A further embodiment of the device according to the invention is shown in fig. 6. In this case, the pressure valve or its opening 4 remains essentially unchanged in comparison with the embodiment shown in fig. 5. The covering means 7 for the suction opening has a multiplicity of apertures 27a, 27b, etc. These apertures 27a, 27b serve to reduce the flow path at the narrowest gap.

Since, in addition to the cross section of the respective gap, the length of the gap between the opening 13 of the valve and the covering means 7 is important, these apertures 27a, 27b can ensure that the gas can discharge directly at the locations which would result in a relatively long gas path of narrow cross section. The respective apertures 27a, 27b may be arranged essentially symmetrically, as in the embodiment shown in fig. 6; however, an arrangement of the respective apertures 27a, 27b which deviates therefrom is also possible. In this embodiment, the opening 13 is of star-shaped design, as a result of which a greatly increased periphery is achieved. The apertures are arranged between the

projections 28a, 28b, etc. of the opening 13.

In the embodiment shown in fig. 7, the covering means 13 has recesses 29a, 29b, etc. instead of the apertures 27a, 27b from fig. 6. In this embodiment, the suction opening 13 likewise has projections 28a, 28b, etc. By means of this measure, firstly the periphery of the suction opening 13 can be greatly increased, and secondly flow paths which are far too long can also be prevented, since the covering means 7 projects slightly beyond the opening in each case only in the region of the respective projections. The projections 39a, 39b, 39c (cf. figs 6 and 7) of the covering means 7 preferably serve to limit the stroke of the covering means.

The advantages of the present invention result from the avoidance of the abovementioned disadvantages (high energetic loss contributions through the suction valve inside the compression device). In particular, it is possible for the passage area of the narrowest valve gap mainly responsible for the pressure loss, or the passage area at the suction valve, to be increased by factors compared with that of the pressure valve.

The compressor power loss for different control efficiencies for the device according to the invention for the exchange of heat is shown in fig. 8. In this case, the sections A, B, C and D again designate the individual ratios for the different control efficiencies 0.8 (A), 0.6 (B), 0.4 (C), 0.2 (D). The power loss in relation to the isentropic power is also plotted here against the respective compressor speed.

It can be seen that, in the device according to the invention for compressing gas, the relative stroke-independent friction power 31a and the relative stroke-dependent friction power 31b also remain essentially constant over the compressor speed range considered. On the other hand, in the device according to the invention for compressing gas, with increasing compressor speed, a considerably smaller increase in the relative suction valve loss compared with the prior art takes place as a function of the compressor speed, in particular at high control efficiencies and high compressor speeds.

On the other hand, at low control efficiencies, the improvement at the suction valve has only a relatively slight effect.

The overall efficiency of an R134a compressor is shown in figures 9 and 10, fig. 9 showing the effective cross section of a compressor according to the prior art and fig. 10 showing that of the device according to the invention. Defined pressures are again taken as a basis here. The respective speed of the compressor is plotted over the x axis, the delivery efficiency is plotted over the y axis and the calculated overall efficiency is plotted over the z axis. It can be seen that, in the case of the device according to the invention, in particular at high delivery efficiencies and high speeds, the overall efficiency is considerably higher than the comparable efficiency in the device according to the prior art. The maximum overall efficiency in the device according to the invention is also markedly higher than

in the device according to the prior art. Whereas for high delivery efficiencies and high compressor speeds the calculated overall efficiency in the device according to the prior art very quickly drops to values below 0.25, the overall efficiency for the device according to the invention is still about 0.35 at the comparable speeds and delivery efficiencies.

As shown in fig. 8, in comparison with fig. 3, it was possible to reduce the suction valve losses to about 30%. In this case, it was possible for the effects on the overall efficiency - shown in figures 8, 9 and 10 - defined as the ratio of isentropic compression power and invested mechanical driving power, in particular at average and high volumetric flows (that is to say average and high speeds and average and high control efficiencies), to be significantly improved, as can be seen from the comparison of figures 9 and 10. The result of this, in effect, is that lower driving power is required for the operation of the air-conditioning system, and in this way the fuel consumption required for the air-conditioning system and the associated emission (greenhouse effect) can be reduced.

In addition, it is possible to reduce the hot-gas temperature. This leads to lower thermal loads on the coolant hoses, to a reduction in the power requirements imposed on the condenser, since less heat is to be dissipated to the environment, and to a reduction in the refrigerant diffusion rate through the elastomeric hose materials, which in turn leads to further protection of the environment.